IN: DiNenno, P.J., et al., Editors, SFPE Handbook of Fire Protection Engineering, 2nd Edition, Chapter 6, Section 3, 3/134-147 pp

ESTIMATING TEMPERATURES IN COMPARTMENT FIRES

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INTRODUCTION

The ability to predict temperatures developed in compartment fires is of great significance to the fire protection professional. There are many uses for a knowledge of compartment fire temperatures, including the prediction of (1) the onset of hazardous conditions, (2) property and structural damage, (3) changes in burning rate, (4) ignition of objects, and (5) the onset of flashover.

The fundamental principles underlying compartment fires are presented in Section 3, Chapter 5. This chapter gives a number of simplified solution techniques.

FIRE STAGES

In this chapter, compartment fires are defined as fires in enclosed spaces, which are commonly thought of as rooms in buildings, but may include other spaces such as those found in transportation vehicles such as ships, planes, trains, and the like.

Compartment fires are often discussed in terms of growth stages.[1] Figure 3-6.1 shows an idealized variation of temperature with time along with the growth stages. The growth stages are:

1. Ignition,
2. Growth,
3. Flashover,
4. Fully developed fire, and
5. Decay.

While many fires will not follow this idealization, it provides a useful framework for the discussion of compartment fires. All fires include an ignition stage but, beyond that, may fail to grow, or they may be affected by manual or automatic suppression activities before going through all of the stages listed above.

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Growth Stage Definitions

Ignition stage: The period during which the fire begins.

Growth stage: Following ignition, the fire initially grows primarily as a function of the fuel itself, with little or no influence from the compartment. The fire can be described in terms of its rate of energy and combustion product generation. A discussion of energy generation or burning rate can be found in Section 3, Chapter 1. If sufficient fuel and oxygen are available, the fire will continue to grow, causing the temperature in the compartment to rise. Fires with sufficient oxygen for combustion are said to be fuel controlled.

Flashover: Flashover is generally defined as the transition from a growing fire to a fully developed fire in which all combustible items in the compartment are involved in fire. During this transition there are rapid changes in the compartment environment. Flashover is not a precise term, and several variations in definition can be found in the literature. Most have criteria based on the temperature at which the radiation from the hot gases in the compartment will ignite all of the combustible contents. Gas temperatures of 300 to 650°C have been associated with the onset of flashover, although temperatures of 500 to 600°C are more widely used.[2] The ignition of unburnt fuel in the hot fire gases, the appearance of flames from openings in a compartment, or the ignition of all of the combustible contents may actually be different phenomena.

Fully developed fire: During this stage, the heat release rate of the fire is the greatest. Frequently during this stage more fuel is pyrolized than can be burned with the oxygen available in the compartment. In this case, the fire is said to be ventilation controlled. If there are openings in the compartment, the unburned fuel will leave the compartment in the gas flow and may burn outside of the compartment. During the fully developed stage, the environment within the compartment has a significant effect on the pyrolysis rate of the burning objects.

Decay stage: Decay occurs as the fuel becomes consumed, and the heat release rate declines. The fire may change from ventilation to fuel controlled during this period.

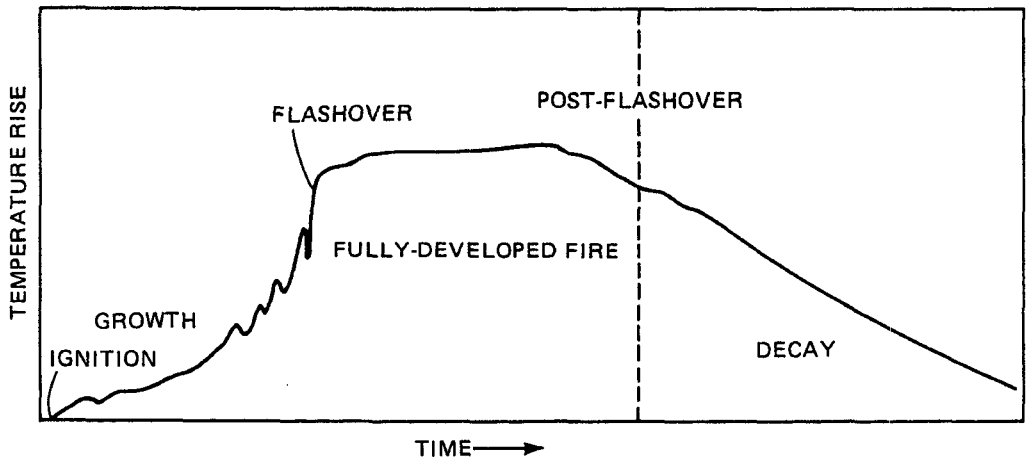

Fig. 3-6.1. General description of room fire in absence of fire control.

COMPARTMENT FIRE PHENOMENA

Compartment Fire Model

In order to calculate or predict the temperatures generated in a compartment fire, a description or model of the fire phenomena must be created. This model will be described in terms of physical equations which can be solved to predict the temperature in the compartment. Such a model is, therefore, an idealization of the compartment fire phenomena. Consider a fire which starts at some point below the ceiling and releases energy and products of combustion. The rate at which energy and products of combustion are released may change with time. The hot products of combustion form a plume, which, due to buoyancy, rises toward the ceiling. As the plume rises, it draws in cool air from within the compartment, decreasing the plume's temperature and increasing its volume flow rate. When the plume reaches the ceiling, it spreads out and forms a hot gas layer which descends with time as the plume's gases continue to flow into it. There is a relatively sharp interface between the hot upper layer and the air in the lower part of the compartment. The only interchange between the air in the lower part of the room and the hot upper layer assumed is through the plume. As the hot layer descends and reaches openings in the compartment walls (e.g., doors and windows), hot gas will flow out the openings and outside air will flow into the openings. This description of compartment fire phenomena is referred to as a two-layer or zone model. The basic compartment fire phenomena are shown schematically in Figure 3-6.2.

The two-layer model concept assumes that the compositions of the layers are uniform. That is, the temperature and other properties are the same throughout each layer. Although the temperature of the lower layer will rise during the course of the fire, the temperature of the upper layer will remain greater and is of the most importance in compartment fires. The assumptions may be less valid for very large spaces or for long, narrow spaces such as corridors and shafts.

Calculation of Compartment Fire Temperatures

The basic principle used to calculate the temperature in a compartment fire is the conservation of energy. As applied to the hot upper layer, the conservation of energy can be simply stated as: the energy added to the hot upper layer by the fire equals the energy lost from the hot layer plus the time rate of change of energy within the hot upper layer. From the time rate of change of energy within the hot layer, the temperature of the layer can be computed. Conservation of energy can also be applied to the lower layer, as well. Since the volume of the upper layer changes with time, and mass flows in and out of the upper layer, conservation of mass must be used along with the conservation of energy. Because the energy generated by the fire and the temperatures in the

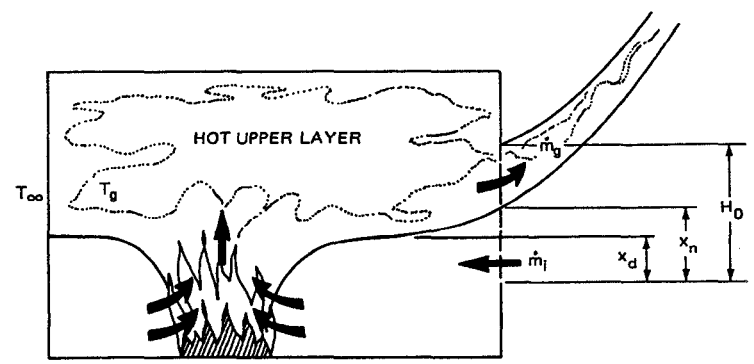

Fig. 3-6.2. Two-layer model with no exchange between layers except the plume.

compartment vary as a function of time, the application of conservation of energy will result in a series of differential equations. For the purposes of examining the components of the conservation of energy, the steady-state expressions for the conservation of energy for the hot upper layer will be used.

The transport of energy in a compartment fire is a very complex process. In order to formulate expressions for the conservation of energy in a practical way, a number of assumptions must be made. It is possible to formulate the equations for the conservation of energy in a number of ways, based on the level of detail desired. The expressions and assumptions used here are based on those commonly found in the fire research literature and represent a somewhat simplified description of the phenomena. Additional details may be found in the references cited.

The steady-state conservation of energy for the hot upper gas layer in a compartment can be simply stated as: the energy generated by the fire and added to the hot layer equals the energy lost from the hot layer through radiation and convection plus the energy convected out the compartment openings.

Energy Generated by the Fire

The energy generated by the fire is the primary influence on the temperature in a compartment fire, and much research has been conducted in predicting the energy release rate of many fuels under a variety of conditions. This discussion will focus on flaming combustion, as it is most important in generating a significant temperature rise in a compartment. A discussion of smoldering combustion is found in Section 2, Chapter 11. As a fuel is heated and releases pyrolysis products, these products react with oxygen, generating heat and producing flames. The rate of energy release is equal to the mass loss rate of the fuel times the heat of combustion of the fuel.

$$\dot{Q} = \dot{m}_f \Delta h_c \qquad (1)$$

where

$\dot{Q}$ = energy release rate of the fire (kW)
$\dot{m}_f$ = mass burning rate of the fuel (kg/s)
Δh_c = effective heat of combustion of the fuel (kJ/kg)

The effective heat of combustion is the heat of combustion which would be expected in a fire where incomplete combustion takes place. This is less than the theoretical heat of combustion as measured in the oxygen bomb calorimeter. The effective heat of combustion is often described as a fraction of the theoretical heat of combustion. The effect of fluctuations is largely neglected.

In fuel-controlled fires, there is sufficient air to react with all the fuel within the compartment. In ventilation-controlled fires, there is insufficient air within the compartment, and some of the pyrolysis products will leave the compartment, possibly to react outside the compartment. For calculating the temperatures produced in compartment fires, the primary interest is in the energy released within the compartment.

The pyrolysis rate of the fuel depends on the fuel type, its geometry, and on the fire-induced environment. The energy generated in the compartment by the burning pyrolysis products then depends on the conditions (temperature, oxygen concentration, etc.) within the compartment. While the processes involved are complex, and some are not well understood, there are two cases where some simplifying assumptions can lead to useful methods for approximation of the energy released by the fire.

Free-burning fires are defined as those in which the pyrolysis rate and the energy release rate are affected only by the burning of the fuel itself and not by the room environment. This is analogous to a fire burning out of doors on a calm day. Babrauskas has provided a collection of data on free-burning fires in Section 3, Chapter 1. This data is most useful for estimating burning rates of primarily horizontal fuels in pre-flashover fires, where the primary heating of the fuel is from the flames of the burning item itself. Vertical fuels, such as wall linings and fuels located in the upper hot gas layer, will likely be influenced by the pre-flashover room environment.

Ventilation-controlled fires are defined as those in which the energy release rate in the room is limited by the amount of available oxygen. The mass flow rate of air or oxygen into the room through a door or window can be calculated from the expressions described below and in Section 2, Chapter 5. For most fuels, the heat released per mass of air consumed is a constant approximately equal to 3000 KJ/kg.[3] Therefore, the rate of energy release of the fire can be approximated from the air inflow rate.

The amount of energy released by the fire which enters the hot upper layer is a function of the fire, layer conditions, and geometry. For most fires, approximately 35 percent of the energy released by the fire leaves the fire plume as radiation.[4] (A discussion of flame radiation can be found in Section 2, Chapter 14.) In a compartment fire, a fraction of the radiated energy reaches the upper layer. The majority of the remaining energy released by the fire is convected into the upper layer by the plume. As the plume rises, it entrains air from the lower layer, thus reducing its temperature and increasing the mass flow rate. For a first approximation, it can be assumed that all of the energy generated by the fire is transported to the upper layer. For a complete discussion of fire plumes see Section 2, Chapter 2.

Conservation of Mass

The mass flow into the compartment and the flow out are related by

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \qquad (2)$$

where

$\dot{m}_f$ = mass burning rate of the fuel (kg/s)

The mass flow rate of hot gas out of a window or door is given by Rockett[5]

$$\dot{m}_g = \frac{2}{3} C_d W_0 \rho_\infty \left[2g \frac{T_\infty}{T_g} \left(1 - \frac{T_\infty}{T_g} \right) \right]^{1/2} (H_0 - X_N)^{3/2} \qquad (3)$$

where

$\dot{m}_g$ = mass flow rate of hot gas out an opening (kg/s)
C_d = orifice constriction coefficient (typically ≈ 0.7)
W_0 = opening width (m)
H_0 = opening height (m)
ρ_∞ = ambient air density (kg/m^2)
g = acceleration due to gravity, 9.8 m/s^2
X_N = height of neutral plane (m)
T_g = temperature of the hot upper gas layer (K)
T_∞ = ambient temperature (K)

The mass flow rate of air into a door or window is given by

$$\dot{m}_a = \frac{2}{3} C_d W_0 \rho_\infty \left[2g\left(1 - \frac{T_\infty}{T_g}\right)\right]^{1/2} (X_N - X_d)^{1/2}(X_N + X_d/2) \quad (4)$$

where

X_d = height of the interface (m)

The expressions for mass flow in and mass flow out cannot be solved directly for T_g, since the height to the neutral plane and interface are unknown. The complete solution of these equations requires expressions for plume entrainment and additional energy equations and is normally carried out only in computer fire models. If the mass burning rate of the fuel is small compared with the mass flow rate of air into the compartment, the mass flow out of the opening may be approximated as equal to the mass inflow rate. Flows out of vents in the ceiling are discussed in Section 3, Chapter 11.

For pre-flashover fires in compartments with typical doors or windows, the neutral plane and interface can be approximated at the midlevel of the opening. This approximation can only be made after the initial smoke filling of the compartment is complete, and flow in and out of the opening is established.

For fires nearing flashover and post-flashover fires the interface between the upper and lower layers is located near the floor, and the flow reaches a maximum for a given upper gas temperature. Rockett has shown the temperature dependence on the flow becomes small above 150°C, and the flow into the compartment can be approximated as a constant times[5]

$$A_0\sqrt{H_0}$$

Rockett calculated values for this constant of 0.40 to 0.61 kg/s · $m^{5/2}$, depending on the discharge coefficient of the opening. Thomas and Heselden estimate the value of this constant at 0.5 kg/s · $m^{5/2}$, which is the value most commonly found in the literature.[6] The resulting approximation is then

$$\dot{m}_a = 0.5\, A_0\sqrt{H_0} \quad (5)$$

where

A_0 = area of opening (m^2)
H_0 = height of opening (m)

The term

$$A_0\sqrt{H_0}$$

is commonly known as the ventilation factor. The first use of this type of opening flow analysis for evaluating post flashover fire test data is attributed to Kawagoe.[7] From early work analyzing such data, the empirical observation was made that wood fires in rooms with small windows appeared to burn at a rate approximately stoichiometric. Although flames emerging from the windows implied that some fuel was burning outside, calculations often suggested that enough air was entering the fire for stoichiometric burning. Empirical observations on wood fires[7] led to

$$\dot{m}_f = 0.09\, A_0\sqrt{H_0} \quad (6)$$

There is now a body of data[8] that modifies this simple proportionality between $\dot{m}_f$ and

$$A_0\sqrt{H_0}$$

The Conseil International du Bâtiment (CIB) experiments upon which Law[9] has based her method shows a dependance on A_T. It seems possible that the wide use of Equation 6 is a result of a concentration of experimental fires in rooms of a limited range of

$$A_T/A_0\sqrt{H_0}$$

where

A_T = total area of the compartment enclosing surfaces (m^2)

Traditionally, energy balances were often stated in terms of the energy produced by the burning fuel and, thereby, led to an effective heat of combustion of the fuel. However, this in principle leads to the same result, the energy produced is related to the air flow for ventilation-controlled fires. Kawagoe[7] and Magnusson and Thelandersson[10] used 10.75 MJ/kg for the effective heat of combustion of wood in the flaming phase for fully developed compartment fires. With 16.4 MJ/kg for the heat of combustion of wood volatiles, this corresponds to a combustion efficiency of 10.75/16.4, which is virtually identical to the 0.65 used in several computer models.

By far the majority of data are based on experiments in which the fuel was cellulosic, and much of the experimental data are based on wood in the form of cribs. For the post flashover burning of a different fuel with a different chemistry, the burning rate expressions may still be used, as long as the fuel is a hydrocarbon producing approximately 3000 kJ for each kg of air consumed in the combustion process. Because different fuels react differently to the thermal environment and will pyrolyze at different rates according to the energy requirements to produce volatiles, one can only estimate temperatures by evaluating the differences, or obtain maximum temperatures by using stoichiometry. Fuels more volatile than wood will probably produce lower temperatures inside a compartment, even if the excess fuel produces a greater hazard outside the compartment. The assumption that the energy is related to the air flow and that the fuel is in stoichiometric proportion will give an upper estimate of temperatures for ventilation-controlled fires. Since Equation 6 is close to stoichiometric, it could, coupled with the effective heat of combustion of wood, give results close to an upper temperature limit for other fuels.

Conservation of Energy

The heat generated by burning materials within a compartment is absorbed by the enclosing surfaces of the compartment and any other structural surfaces, and by the surfaces of the fuel and by the incoming air and any excess fuel. Heat is lost to the exterior in the flames and hot gases that exit from the openings in the compartment enclosing surfaces and by radiation through the openings. An example of an experimental heat balance measured in a small compartment is given in Table 3-6.1. For this compartment, unglazed windows provided ventilation from the start of the fire.

Table 3-6.1 illustrates the significant amount of heat loss in the effluent gases and shows that, with decreasing

TABLE 3-6.1 ***Heat Balance Measured in Experimental Fires in a Compartment of 29 m² Floor Area with a Fire Load of Wood Cribs***

Fire load (kg)	Window area (m^2)	Heat release (kcal/s)	Heat loss from hot gases (%)			
			Effluent gas	Structural surfaces	Feedback to fuel	Window radiation
877	11.2	1900	65	15	11	9
	5.6	1900	52	26	11	11
1744	11.2	3200	61	15	11	13
	5.6	2300	53	26	12	9
	2.6	1600	47	30	16	7

window area, a larger proportion of the heat released will be absorbed by the enclosing surfaces. The total heat released, assuming a complete burnout, is directly proportional to the amount of the fire load, but the rate of heat release may also be controlled by the ventilation. In this example, with the lower fire load, both window areas give sufficient ventilation for the fuel to burn at its maximum (free burning) rate but, with the doubled fire load, the burning rate is not doubled, because the window area restricts the ventilation needed.

METHODS FOR PREDICTING PRE-FLASHOVER COMPARTMENT FIRE TEMPERATURES

The solution of a relatively complete set of equations for the conservation of energy requires the solution of a large number of equations which vary with time. Although individual energy transport equations may be solved, in general there is not an explicit solution for a set of these equations. As a result, one of two approaches can be taken. The first is an approximate solution which can be accomplished by "hand" using a limiting set of assumptions. The second is a more complete solution utilizing a computer program. In either case, a number of methods have been developed. The methods presented are those which appear most widely accepted in the fire protection community. Each method employs assumptions and limitations which should be understood before employing the method. The methods presented in this chapter predict average temperatures and are not applicable to cases where prediction of local temperatures are desired. For example, these methods should not be used to predict detector or sprinkler actuation or the temperatures of materials as a result of direct flame impingement.

Method of McCaffrey, Quintiere, and Harkleroad

McCaffrey, Quintiere, and Harkleroad have used a simple conservation of energy expression and a correlation with data to develop an approximation of the upper layer temperature in a compartment.[11] Applying the conservation of energy to the upper layer yields

$$\dot{Q} = \dot{m}_g c_p (T_g - T_\infty) + q_{loss} \tag{7}$$

where

$\dot{Q}$ = energy (heat) release rate of the fire (kW)
$\dot{m}_g$ = gas flow rate out the opening (kg/s)
c_p = specific heat of gas (kJ/kg · K)
T_g = temperature of the upper gas layer (K)
T_∞ = ambient temperature (K)
q_{loss} = net radiative and convective heat transfer from the upper gas layer (kW)

The left-hand side of Equation 7 is the energy generated by the fire. On the right-hand side, the first term is the heat transported from the upper layer in the gas flow out an opening. The second term is the net rate of radiative and convective heat transfer from the upper layer, which is approximately equal to rate of heat conduction into the compartment surfaces. The rate of heat transfer to the surfaces is approximated by

$$q_{loss} = h_k A_T (T_g - T_\infty) \tag{8}$$

where

h_k = effective heat transfer coefficient (kW/m · K)
A_T = total area of the compartment enclosing surfaces (m^2)

Substituting Equation 8 into Equation 7 yields the nondimensional temperature rise in terms of two dimensionless groups

$$\frac{\Delta T_g}{T_\infty} = \frac{\dot{Q}/(c_p T_\infty \dot{m}_g)}{1 + h_k A_T/(c_p \dot{m}_g)} \tag{9}$$

where

ΔT_g = upper gas temperature rise above ambient ($T_g - T_\infty$) (K).

The mass flow rate of hot gas out of a window or door can be rewritten from Equation 3.

$$\dot{m}_g = \frac{2}{3} C_d W_0 H_0^{3/2} \rho_\infty \left[2g \frac{T_\infty}{T_g} \left(1 - \frac{T_\infty}{T_g} \right) \right]^{1/2} \left(1 - \frac{X_N}{H_0} \right)^{3/2} \tag{10}$$

where

C_d = orifice constriction coefficient
W_0 = opening width (m)
H_0 = opening height (m)
ρ_∞ = ambient air density (kg/m^2)
g = acceleration due to gravity, 9.8 m/s^2
X_N = height of neutral plane (m)

Since X_N primarily depends on T_g, $\dot{Q}$, and geometric factors (H_0 and W_0), $\dot{m}_g$ may be replaced by

$$\sqrt{g}\, \rho_\infty A_0 \sqrt{H_0}$$

in the two dimensionless variables in Equation 10, without any loss in generality. The effects of T_g and $\dot{Q}$ are incorporated into the correlation via other terms. Based on an analysis of test data, Equation 9 was written as a power-law relationship

$$\Delta T_g = 480\left(\frac{\dot{Q}}{\sqrt{g}c_p\rho_\infty T_\infty A_0\sqrt{H_0}}\right)^{2/3}\left(\frac{h_k A_T}{\sqrt{g}c_p\rho_\infty A_0\sqrt{H_0}}\right)^{-1/3} \quad (11)$$

where

A_0 = area of opening (m^2)
H_0 = height of opening (m)

The numbers 480, ⅔, and $-⅓$ were determined by correlating the expression with the data from over 100 experimental fires. These data included both steady-state and transient fires in cellulosic and synthetic polymeric materials and gaseous hydrocarbon fuels. Compartment height ranged from 0.3 m to 2.7 m and floor areas from 0.14 m^2 to 12.0 m^2. The compartments contained a variety of window and door sizes. The term raised to the ⅔ power in Equation 11 represents the ratio of the energy released to the energy convected, and the term raised to the $-⅓$ power represents the energy lost divided by the energy convected.

Substituting the values for ambient conditions of

g = 9.8 m/s^2
c_p = 1.05 kJ/kg · K
ρ_∞ = 1.2 kg/m^3
T_∞ = 295 K

into Equation 11 yields[12,13]

$$\Delta T_g = 6.85\left(\frac{\dot{Q}^2}{A_0\sqrt{H_0}h_k A_T}\right)^{1/3} \quad (12)$$

The heat transfer coefficient can be determined using a steady-state approximation when the time of exposure, t, is greater than the thermal penetration time, t_p, by

$$h_k = k/\delta \qquad \text{for } t > t_p \quad (13)$$

The thermal penetration time is defined as

$$t_p = (\rho c/k)(\delta/2)^2 \quad (14)$$

where

ρ = density of the compartment surface (kg/m^3)
c = specific heat of the compartment surface material (kJ/kg · K)
k = thermal conductivity of compartment surface (kW/m · K)
δ = thickness of compartment surface (m)
t = exposure time (s)
t_p = thermal penetration time (s)

When the time of exposure is less than the penetration time, an approximation based on conduction in a semi-infinite solid is

$$h_k = (k\rho c/t)^{1/2} \qquad \text{for } t \le t_p \quad (15)$$

If there are several wall and/or ceiling materials in the compartment, an area-weighted average for h_k should be used.

The limitations as stated by McCaffrey *et al* on the use of this method for estimating temperatures are:

1. The correlation holds for compartment upper layer gas temperatures up to approximately 600°C,
2. It applies to steady-state as well as time-dependent fires, provided the primary transient response is the wall conduction phenomenon,
3. It is not applicable to rapidly developing fires in large enclosures in which significant fire growth has occurred before the combustion products have exited the compartment,
4. The energy release rate of the fire must be determined from data or other correlations,
5. The characteristic fire growth time and thermal penetration time of the room-lining materials must be determined in order to evaluate the effective heat transfer coefficient, and
6. The correlation is based on data from a limited number of experiments and does not contain extensive data on ventilation-controlled fires nor data on combustible walls or ceilings. Most of the fuel in the test fires was near the center of the room.

Example of McCaffrey *et al* method: Calculate the upper layer temperature of a room 3 × 3 m in floor area and 2.4 m high with a door opening 1.8 m high and 0.6 m wide. The fire source is a steady 750 kW fire. The wall lining material is 0.016 m (⅝ in.) gypsum plaster on metal lath. Perform the calculation at times of 10, 60, and 600 seconds after ignition. Using Equation 11

$$\Delta T_g = 480\left(\frac{\dot{Q}}{\sqrt{g}c_p\rho_\infty T_\infty A_0\sqrt{H_0}}\right)^{2/3}\left(\frac{h_k A_T}{\sqrt{g}c_p\rho_\infty A_0\sqrt{H_0}}\right)^{-1/3}$$

where

c_p = 1 kJ/kg · K
T_∞ = 27°C (300 K)
ρ_∞ = 1.18 kg/m^3
A_0 = 1.8 m × 0.6 m = 1.08 m^2
g = 9.8 m/s^2
H_0 = 1.8 m
$\dot{Q}$ = 750 kW
$A_T = A_{walls} + A_{floor} + A_{ceiling} - A_{openings}$
$= 4 \times (3 \times 2.4) + (3 \times 3) + (3 \times 3) - 1.08$
$= 28.8\ m^2 + 9\ m^2 + 9\ m^2 - 1.08$
$= 45.72\ m^2$

The wall heat loss coefficient, h_k, is a function of time.

a. Calculate the thermal penetration time, t_p.

$$t_p = (\rho c/k)(\delta/2)^2$$

where

ρ = wall material density (1440 kg/m^3)
k = 0.48 × 10^{-3} kW/m · c
c = 0.84 kJ/kg°C
δ = 0.016 m
t_p = 161.3 s

b. Calculate h_k at 10, 60, and 600 s.

For $t < t_p$ (10, 60 s)

$$h_k = (k\rho c/t)^{1/2} \qquad k\rho c = 0.581$$

1. At t = 10 s.

$$h_k = (0.581/10)^{1/2} = 0.24\ \text{kW/m} \cdot \text{K}$$

2. At $t = 60$ s.

$$h_k = (0.581/60)^{1/2} = 0.098 \text{ kW/m} \cdot \text{K}$$

3. For $t > t_p$ (600 s) at $t = 600$ s.

$$h_k = k/\delta = 0.48 \times 10^{-3}/0.016 = 0.03 \text{ kW/m} \cdot \text{K}$$

c. Calculate the compartment temperature at the three times using Equation 11.

1. At $t = 10$ s.

$$\Delta T_g = 480\left[\frac{750}{(\sqrt{9.8})(1)(300)(1.08)(\sqrt{1.8})}\right]^{2/3} \cdot \left[\frac{(0.24)(45.72)}{(\sqrt{9.8})(1)(1.18)(1.08)(\sqrt{1.8})}\right]^{-1/3}$$

$$= 480(0.55)^{2/3}(2.05)^{-1/3}$$

$$\Delta T_g = 254 \text{ K}$$

2. At $t = 60$ s.

$$\Delta T_g = (480)(0.55)^{2/3}(0.837)^{-1/3}$$

$$\Delta T_g = 342 \text{ K}$$

3. At $t = 600$ s.

$$\Delta T_g = (480)(0.55)^{2/3}(0.26)^{-1/3}$$

$$\Delta T_g = 506 \text{ K}$$

Method of Foote, Pagni and Alvares

This method follows the basic correlations of McCaffrey, Quintiere, and Harkleroad and adds data for forced-ventilation fires. Using Equation 9 and not introducing an expression for doorway flow results in the expression[14]

$$\frac{\Delta T_g}{T_\infty} = 0.63\left(\frac{\dot{Q}}{\dot{m}c_p T_\infty}\right)^{0.72}\left(\frac{h_k A_T}{\dot{m}c_p}\right)^{-0.36} \quad (16)$$

where

ΔT_g = upper gas temperature rise above ambient (K)
T_∞ = ambient air temperature (K)
$\dot{Q}$ = energy (heat) release rate of the fire (kW)
$\dot{m}$ = compartment mass ventilation rate (kg/s)
c_p = specific heat of gas (kJ/kg · K)
h_k = effective heat transfer coefficient (kW/m · K)
A_T = total area of the compartment-enclosing surfaces (m^2)

The coefficient and exponents are based on data from well-ventilated tests in a compartment with a 6 × 4 m floor area and a height of 4.5 m with ventilation rates of 110 to 325 g/s. The compartment exhaust was through a 0.65 × 0.65 m duct located 3.6 m above the floor. Four air inlet openings were 0.5 × 0.12 m high, with centerlines 0.1 m above the floor. A methane gas burner fire in the center of the floor with heat release rates of 150 to 490 kW resulted in upper gas temperatures of approximately 100 to 300°C.

Foote *et al* have shown that the correlation for forced-ventilation fires agrees well with the data presented by McCaffrey *et al* for free ventilation fires with

$$\dot{m} \approx 0.1(\rho_\infty \sqrt{g} A_0 \sqrt{H_0})$$

Example of Foote *et al* method: Estimate the temperature in a 5 × 5 m in floor area × 4 m high compartment having 0.025-m-(1-in.)-thick concrete walls. The forced-ventilation rate is 2.4 m^3/s of air (5000 cfm). Perform the calculation for $t > t_p$. The fire size is given as 1000 kW; ambient air conditions at 300 K. Using Equation 16

$$\frac{\Delta T_g}{T_\infty} = 0.63(\dot{Q}/\dot{m}c_p T_\infty)^{0.72}(h_k A_T/\dot{m}c_p)^{-0.36}$$

where

$\dot{Q}$ = 1000 kW
T_∞ = 300 K
c_p = 1.0 kJ/kg · K
A_T = 4 × (5 × 4) + 2 (5 × 5) = 105 m^2
$\dot{m}$ = (2.4 m^3/s) (1.18 kg/m^3) = 2.8 kg/s

Calculate h_k for $t > t_p$. For 0.025-m-thick concrete

δ = 0.025 m
ρ = 2000 kg/m^3
k = 1.4 × 10^{-3} kW/m · K
c_p = 0.88 kJ/kg · K

$$t_p = (\rho c/k)(\delta/2)^2$$

$$= \left[\frac{(2000)\cdot(0.88)}{1.4\times10^{-3}}\right]\left(\frac{0.025}{2}\right)^2$$

$$= 196 \text{ s} \quad \text{for } t > t_p$$

$$h_k = k/\delta = 1.4 \times 10^{-3}/0.025 = 0.056 \text{ kW/m}^2 \cdot \text{K}$$

$$\frac{\Delta T_g}{T_\infty} = (0.63)\left[\frac{1000}{(2.8)(1)(300)}\right]^{0.72}\left[\frac{(0.056)(105)}{(2.8)(1)}\right]^{-0.36}$$

$$\Delta T_g = (0.14)(T_\infty)$$

$$\Delta T_g = 164 \text{ K}$$

$$T_g = 164 + 300 \text{ K} = 464 \text{ K}$$

METHODS FOR PREDICTING POST FLASHOVER COMPARTMENT FIRE TEMPERATURES

Method of Babrauskas

The following method is based on the work of Babrauskas.[15,16] The upper gas temperature, T_g, is expressed according to a series of factors, each one accounting for a different physical phenomenon

$$T_g = T_\infty + (T^* - T_\infty) \cdot \theta_1 \cdot \theta_2 \cdot \theta_3 \cdot \theta_4 \cdot \theta_5 \quad (17)$$

where T^* is an empirical constant = 1725 K, and the factors θ are in Equations 23, 28, 30, 31, 33, and 34.

Burning rate stoichiometry, θ_1: The dimensionless stoichiometric coefficient ϕ is defined as

$$\phi = \frac{\dot{m}_f}{\dot{m}_{f,st}} \quad (18)$$

where $\dot{m}$ is the fuel mass pyrolysis rate (kg/s) and $\dot{m}_{f,st}$ is the stoichiometric mass burning rate (i.e., no excess fuel and no excess oxygen).

$$\dot{m}_{f,st} = \frac{0.5 A_0 \sqrt{H_0}}{r} \quad (19)$$

where the ratio r is such that 1 kg fuel + r kg air → (1 + r) kg products. The value of r is readily computable for fuels containing carbon, hydrogen, and/or oxygen from the chemical formula of the fuel, taking the products to be CO_2, H_2O, and N_2.

$$C_xH_yO_z + wO_2 + w\left(\frac{79}{21}\right)N_2 \rightarrow xCO_2 + \frac{y}{2}H_2O + w\left(\frac{79}{21}\right)N_2 \quad (20)$$

where

$$w = \frac{2x + \frac{y}{2} - z}{2} \quad (21)$$

and

$$r = \frac{[w + w(3.76)]28.97}{12.01x + 1.00y + 16.00z} \quad (22)$$

At stoichiometry $\phi = 1$, and it is greater than 1 for fuel-rich burning and less than 1 for fuel-lean conditions.

The effect of ϕ on gas temperatures was evaluated by numerical computations using the COMPF2 computer program.[17] The efficiency factor, θ_1, accounts for deviation from stoichiometry and is shown in Figure 3-6.3. It is seen that the fuel-lean and the fuel-rich regimes exhibit a very different dependence. For the fuel-lean regime, the results can be approximated by

$$\theta_1 = 1.0 + 0.51 \ln \phi \qquad \text{for } \phi < 1 \quad (23)$$

Similarly, in the fuel-rich regime a suitable approximation is

$$\theta_1 = 1.0 - 0.05(\ln \phi)^{5/3} \qquad \text{for } \phi > 1 \quad (24)$$

If heat release rate, $\dot{Q}$, rather than mass loss rate, $\dot{m}$, is used, then

$$\phi = \frac{\dot{Q}}{\dot{Q}_{st}} \quad (25)$$

And, since the stoichiometric heat release rate is

$$\dot{Q} = 1500\, A_0\sqrt{H_0} \quad (26)$$

then

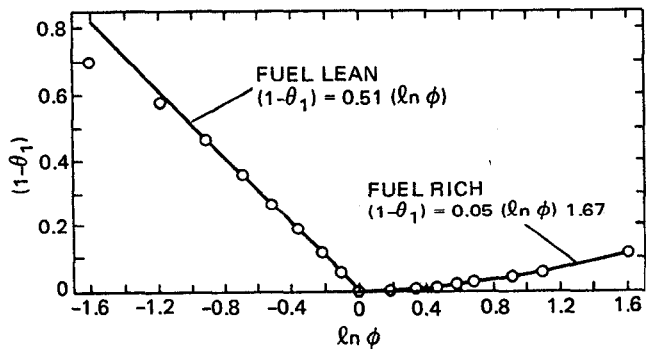

Fig. 3-6.3. Effect of equivalence ratio.

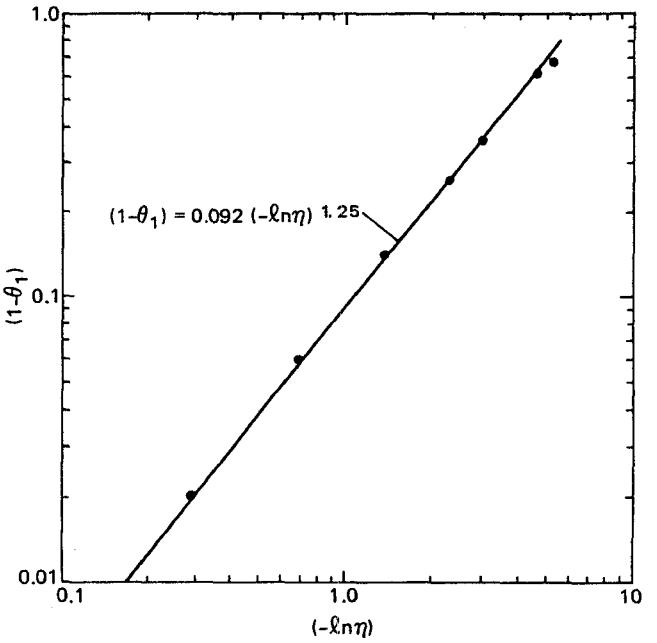

Fig. 3-6.4. Effect of pool diameter.

$$\phi = \frac{\dot{Q}}{1500 A_0\sqrt{H_0}} \quad (27)$$

The value of $\dot{Q}$ can be determined from Section 3, Chapter 1.

A separate procedure is necessary for pool fires, due to the strong radiative coupling. Here

$$\theta_1 = 1.0 - 0.092(-\ln \eta)^{1.25} \quad (28)$$

where

$$\eta = \left(\frac{A_0\sqrt{H_0}}{A_f}\right)\frac{0.5\Delta h_p}{r\sigma(T_g^4 - T_b^4)} \quad (29)$$

where

Δh_p = heat of vaporization of liquid (kJ/kg)
A_f = pool area (m^2)
σ = Stefan-Boltzmann constant (5.67×10^{-11} kW/$m^2 \cdot K^4$)
T_b = liquid boiling point (K)

This expression unfortunately requires an estimate for T_g to be made, so for the pool fire case, a certain amount of iteration is necessary. The relationship above is plotted in Figure 3-6.4.

Wall steady-state losses, θ_2: The next efficiency factor, θ_2, accounts for variable groups of importance involving the wall surface (which is defined to include the ceiling) properties: area A_T (m^2), thickness L (m), density ρ (kg/m^3), thermal conductivity k (kW/m · K), and heat capacity c_p (kJ/kg · K). This factor is given as

$$\theta_2 = 1.0 - 0.94 \exp\left[-54\left(\frac{A_0\sqrt{H_0}}{A_T}\right)^{2/3}\left(\frac{L}{k}\right)^{1/3}\right] \quad (30)$$

and is shown in Figure 3-6.5.

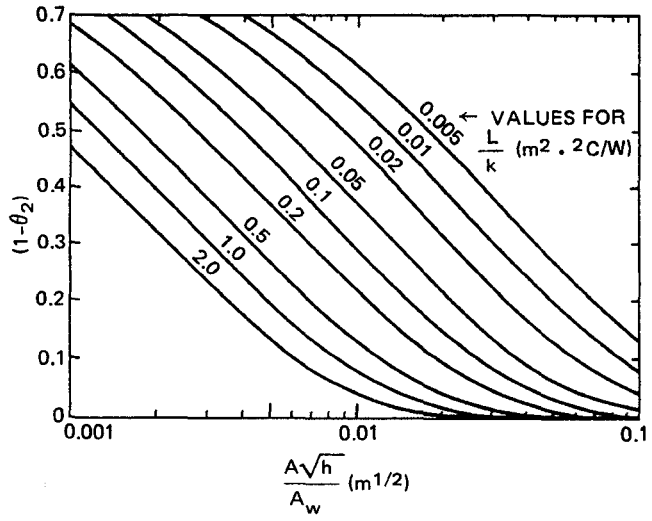

Fig. 3-6.5. Effect of wall steady-state losses.

Wall transient losses, θ_3: For the transient case, the above relationship predicts the asymptotic temperature value. An additional time-dependent factor, however, is needed. See Figure 3-6.6.

$$\theta_3 = 1.0 - 0.92 \exp\left[-150\left(\frac{A_0\sqrt{H_0}}{A_T}\right)^{0.6}\left(\frac{t}{k\rho c_p}\right)^{0.4}\right] \qquad (31)$$

If only steady-state temperatures need to be evaluated, then $\theta_3 = 1.0$.

Wall effects for t just slightly greater than zero are not well modeled with the above relationships for $\theta_2 \times \theta_3$; however, this is not a serious limitation, since the method is only designed for post-flashover fires.

For transient fires, the possibility of two separate effects must be considered. First, the wall loss effect, represented by Equation 31, in all fires, exhibits a non-steady character. Second, the fuel release rate may not be constant. Since in the calculational procedure the previous results are not stored, it is appropriate to restrict consideration to fires where $\dot{m}_f$ does not change drastically over the time scale

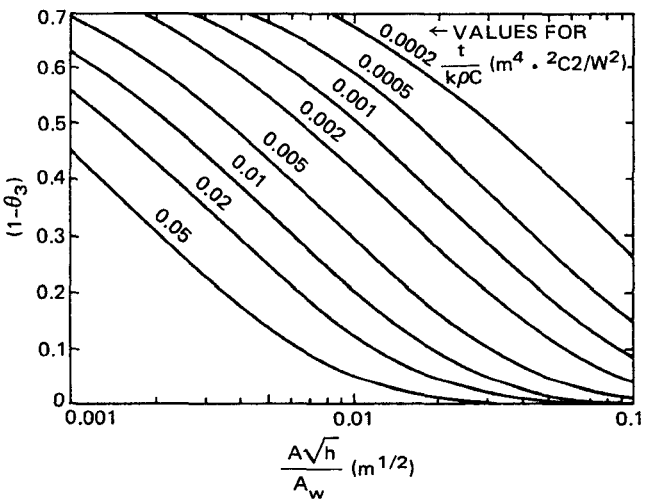

Fig. 3-6.6. Effect of wall transient losses.

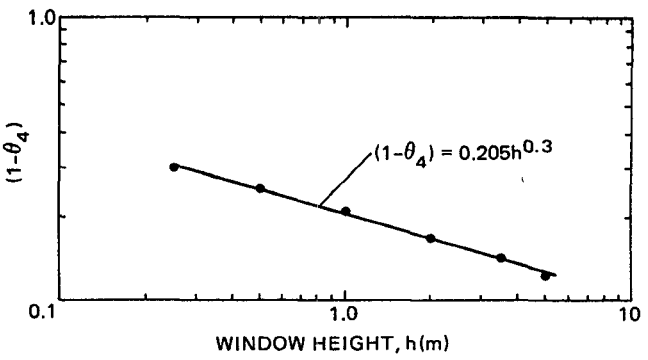

Fig. 3-6.7. Effect of window height.

established by θ_3. This "natural" time scale can be determined as the time when the response has risen to 63 percent of its ultimate value, i.e., at $\theta_3 = 0.63$, and is

$$t = 2.92 \times 10^{-6}(k\rho c_p)\left(\frac{A_T}{A_0\sqrt{H_0}}\right)^{1.5} \qquad (32)$$

Opening height effect, θ_4: The normalization of burning rate and wall loss quantities with the ventilation factor

$$A_0\sqrt{H_0}$$

does not completely determine the total heat balance. An opening of a given

$$A_0\sqrt{H_0}$$

can be tall and narrow or short and squat. For the shorter opening, the area will have to be larger. Radiation losses are proportional to the opening area and will, therefore, be higher for the shorter opening. By slight simplification, a representation for θ_4 can be made as

$$\theta_4 = 1.0 - 0.205H_0^{-0.3} \qquad (33)$$

as shown in Figure 3-6.7.

Combustion efficiency, θ_5: The fire compartment is viewed as a well, but not perfectly, stirred reactor. Thus a certain "unmixedness" is present. A maximum combustion efficiency, b_p, can be used to characterize this. Since the model assumes infinitely fast kinetics, any limitations can also be included here. Data have not been available to characterize b_p in real fires, but agreement with measured fires can generally be obtained with b_p values in the range 0.5 to 0.9. The effect of b_p variation can be described by

$$\theta_5 = 1.0 + 0.5 \ln b_p \qquad (34)$$

as shown in Figure 3-6.8.

Method of Law

The area of structural surface to which heat is lost is expressed by $(A_T - A_0)$. For a given fire load, compartments with different values of A_T, A_0, and height H_0 will have a different heat balance, and thus the temperatures in the compartments will differ. This is illustrated in Figure 3-6.9 which shows how temperature varies with

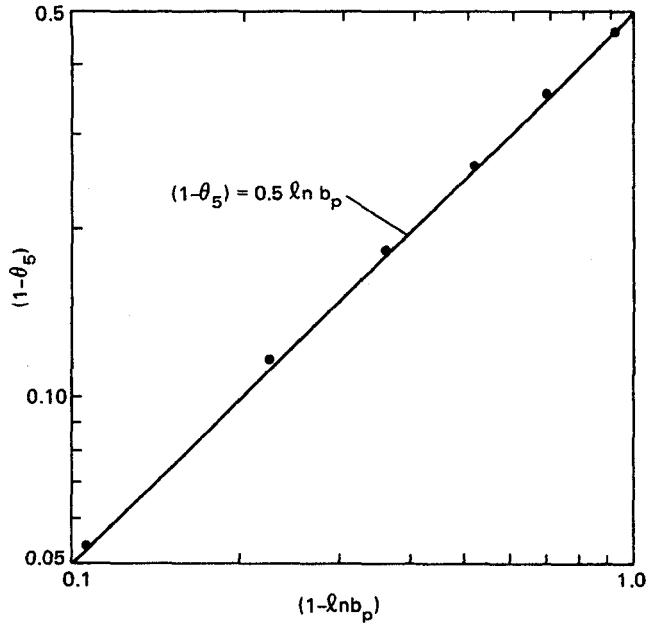

Fig. 3-6.8. Effect of b_p, the maximum combustion efficiency.

$$\Omega = (A_T - A_0)/A_0\sqrt{H_0}$$

For low values of Ω (i.e., high ventilation), the rate of heat release is at a maximum, but the heat loss from the window is also large and the resultant temperature is low. For high values of Ω (i.e., low-ventilation areas), there is little heat loss to the outside, but the rate of heat release is also small and the resultant temperature is, again, low.

The curve in Figure 3-6.10 has been derived from many experimental fires conducted internationally by CIB.[8] For design purposes, Law has defined it as follows

$$T_{g(max)} = 6000\frac{(1-e^{-0.1\Omega})}{\sqrt{\Omega}} \qquad (35)$$

where

$$\Omega = \frac{(A_T - A_0)}{A_0\sqrt{H_0}}$$

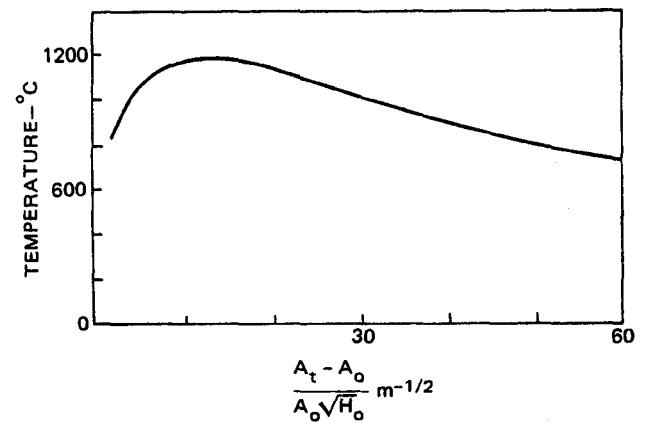

Fig. 3-6.9. Average temperature during fully developed period measured in experimental fires in compartments.

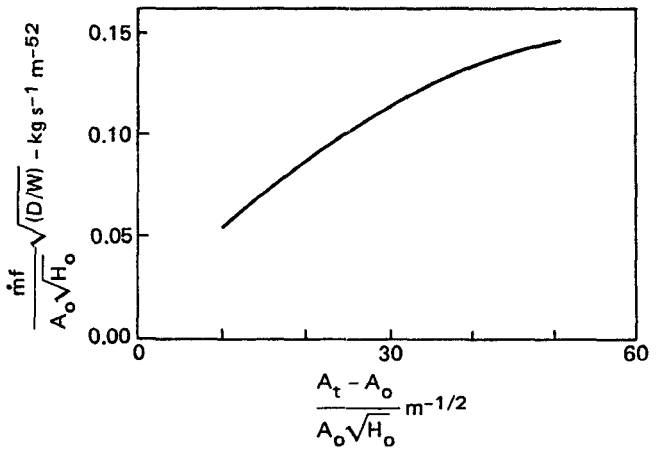

Fig. 3-6.10. Variation of a rate of burning during fully developed period measured in experimental fires in compartments.

and

A_T = total area of the compartment enclosing surfaces (m^2)
A_0 = area of opening (m^2)
H_0 = height of opening (m)

This represents an upper limit of fire temperature rise for a given Ω. However, if the fire load is low, this value may not be obtained. The importance of the effect of fire load also depends on A_0 and A_T, and can be expressed as

$$T_g = T_{g(max)}(1 - e^{-0.05\psi}) \qquad (36)$$

where

$$\psi = \frac{L}{[A_0(A_T - A_0)]^{0.5}}$$

where

L = fire load (wood) (kg)

The effect of the fire on the structure depends not only on the value of T_g but also on the duration of heating. The effective fire duration, τ, in seconds, is given by

$$\tau = \frac{L}{\dot{m}_f} \qquad (37)$$

where

$\dot{m}_f$ = rate of burning kg/s

Equation 6 implies that the smaller the value of

$$A_0\sqrt{H_0}$$

the lower the rate of burning and the longer the duration. Assuming a complete burnout, therefore, the effect on the structure tends to be more severe for large values of Ω

$$\text{small } A_0\sqrt{H_0}$$

For design purposes the following equation has been developed to express the correlation of experimental results[9]

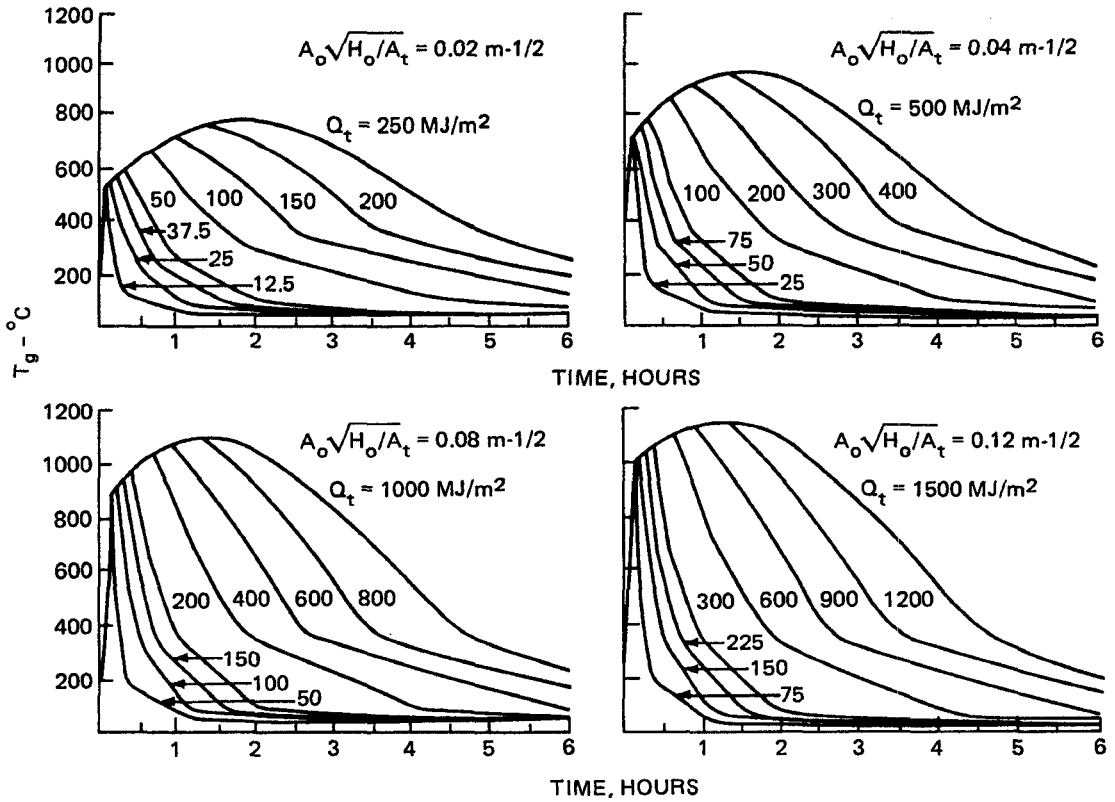

Fig. 3-6.11. Examples of gas temperature-time curves of post-flashover compartment fires for different values of the fire load density Q_t MJ per unit of total internal surface area A_t and the opening factor $A_0\sqrt{H_0/A_t}$. Fire compartment, type A—from authorized Swedish Standard Specifications.[10]

$$\dot{m}_f = 0.18A_0\sqrt{H_0}(W/D)(1 - e^{-0.036\Omega}) \quad \xi < 60 \qquad (38)$$

where

W = compartment width (m)
D = compartment depth (m)

$$\xi = \frac{\dot{m}_f}{A_0\sqrt{H_0}}\left(\frac{D}{W}\right)^{1/2}$$

Equation 38 is shown in Figure 3-6.10 over the range where the data lie. Both equations are for ventilation-controlled fires. When there is ample ventilation, so that the fuel is free burning, the value of $\dot{m}_f$ depends on L and the type of fuel. For example, domestic furniture has a free-burning fire duration of about 20 min, giving τ = 1200 s and $\dot{m}_f = L/1200$.

The temperatures discussed above are averages measured during the fully developed period of the fire. It is assumed that all fires are ventilation controlled, with the simple relationship for rate of burning given by Equation 38, which is near stoichiometric burning, and it is assumed that combustion of 1 kg of wood releases 18.8 MJ in total.

Swedish Method

This method, developed by Magnusson and Thelandersson,[10] is based on the conventional mass and energy balance equations. The fire itself is not modeled; heat release rate curves are provided as input and, in all instances, the energy release must be less than stoichiometric. The method does not take into account that the actual mass loss rate may be greater than stoichiometric, with the excess fuel burning outside the compartment. A computer program SFIRE (versions 1 through 3) is available to perform this method. The results from the computer program have been compared with a large number of full-scale fire experiments, both in the fuel- and ventilation-controlled regimes, with good agreement between theory and experiment. It should be added, however, that most of the experiments involved wood crib fires, which inherently burn slower and produce less excess fuel load than furnishings and other combustibles found in practical fire loads. In the Swedish method, the fire load is expressed in relation to A_T as $\dot{Q}$ 18.8 L/A_T MJ/m².

The design curves approved by the Swedish authorities were computed on the basis of systemized ventilation-controlled heat-release curves taken from reference.[10] Figure 3-6.11 shows some typical curves. The curves are calculated for wall, floor, and ceiling materials with "normal" thermal properties from an energy balance which assumes a uniform temperature in the compartment.

PREDICTING FLASHOVER

One of the uses of predicted compartment fire temperatures is the estimation of the likelihood of flashover. The methods used are similar to those used in the prediction of temperature. In one case, that of McCaffrey *et al*, the method is simply an extension of the temperature calculation.

Method of Babrauskas

Babrauskas uses the energy balance for the upper layer given in Equation 7, where the gas flow rate out of the opening is approximated by[18]

$$\dot{m}_g \approx 0.5A_0\sqrt{H_0} \tag{39}$$

The primary energy loss is assumed to be radiation to 40 percent of the wall area which is at approximately ambient temperature

$$q_{loss} = \varepsilon\sigma(T_g^4 - T_\infty^4)(0.40A_T) \tag{40}$$

where

ε = emissivity of the hot gas
σ = Stefan-Boltzmann constant 5.67×10^{11} kW/m$^2 \cdot$ K^4

Combining Equations 7, 39, and 40, using a gas temperature for flashover of 873 K, a specific heat of air of 1.0 kJ/kg · K, an emissivity of 0.5, and assuming the correlation between compartment wall and opening area of

$$A_T/A_0\sqrt{H_0} \approx 50$$

yields a minimum $\dot{Q}$ required for flashover

$$\dot{Q} = 600A_0\sqrt{H_0} \tag{41}$$

The air flow into the compartment has been approximated as

$$0.5A_0\sqrt{H_0}$$

The maximum amount of fuel which can be burned completely with this air is known as the stoichiometric amount. For most fuels, the heat released per mass of air consumed is a constant approximately equal to 3000 KJ/kg. Therefore, the stoichiometric heat release rate $\dot{Q}_{stoich}$ can be calculated

$$\begin{aligned}\dot{Q}_{stoich} &= 3000\,\dot{m}_g = 3000(0.5A_0\sqrt{H_0}) \\ &= 1500A_0\sqrt{H_0}\end{aligned} \tag{42}$$

From this derivation, it is shown that the minimum $\dot{Q}$ required for flashover equals 0.4 $\dot{Q}_{stoich}$. Comparing these results with fire tests, Babrauskas found that the data falls within a range of $\dot{Q} = 0.3\,\dot{Q}_{stoich}$ to $\dot{Q} = 0.7\,\dot{Q}_{stoich}$. A best fit of the data suggests

$$\dot{Q} = 0.5\dot{Q}_{stoich}$$

which, substituting into Equation 42 yields

$$\dot{Q} = 750A_0\sqrt{H_0} \tag{43}$$

The 33 test fires used had energy release rates from 11 to 3840 kW, with fuels primarily of wood and polyurethane. Ventilation factors

$$A_0\sqrt{H_0}$$

ranged from 0.03 to 7.51 m$^{5/2}$, and surface area to ventilation factor ratios

$$A_T/A_0\sqrt{H_0}$$

ranged from 9 to 65 m$^{-1/2}$.

Example of Babrauskas' method: Calculate the heat release rate necessary to cause flashover, using the method of Babrauskas. Assume the same room as in the McCaffrey *et al* method example for predicting compartment fire temperatures. From Equation 43

$$\dot{Q} = 750A_0\sqrt{H_0}$$

where

A_0 = 1.08 m^2
H_0 = 1.8 m
$\dot{Q}$ = (750) (1.08) $(1.8)^{1/2}$ = 1087 kW

Method of McCaffrey, Quintiere, and Harkleroad

The method of McCaffrey, Quintiere, and Harkleroad for predicting compartment fire temperatures may be extended to predict the energy release rate of the fire required to result in flashover in the compartment.

Equation 11 can be rewritten as

$$\dot{Q} = \left[\sqrt{g}c_p\rho_\infty T_\infty^2\left(\frac{\Delta T_g}{480}\right)^3\right]^{1/2}(h_k A_T A_0\sqrt{H_0}) \tag{44}$$

Selecting an upper gas temperature of 522°C and ambient temperature of 295 K or ΔT_g = 500°C for flashover, and substituting values for the gravitational constant (g = 9.8 m/s^2), the specific heat of air (c_p = 1.0 kJ/kg · K), and the density of air (ρ_∞ = 1.18 kg/m^3), and rounding 607.8 to 610 yields

$$\dot{Q} = 610(h_k A_T A_0\sqrt{H_0})^{1/2} \tag{45}$$

where

h_k = effective heat transfer coefficient [(kW/m)/K]
A_T = total area of the compartment surfaces (m^2)
A_0 = area of opening (m^2)
H_0 = height of opening (m)

Using Equation 12 yields a slightly different value, 623.6 rounded to 620, of the leading coefficient because of the difference in the value used for the specific heat of air

$$\dot{Q} = 620(h_k A_T A_0\sqrt{H_0})^{1/2} \tag{46}$$

The use of either 610 or 620 is acceptable within the accuracy of the expression.

Example of McCaffrey *et al* method: Estimate the energy release rate required for flashover of a compartment. Assume the same room as in the McCaffrey *et al* method example for predicting compartment fire temperatures. Assuming ΔT_g = 500°C as a condition for flashover, and air properties at 295 K, use Equation 45

$$\dot{Q} = 610(h_k A_T A_0\sqrt{H_0})^{1/2}$$

where

$h_k = k/\delta = 0.48 \times 10^{-3}/0.016 = 0.03$ kW/m · K
A_T = 45.72 m^2

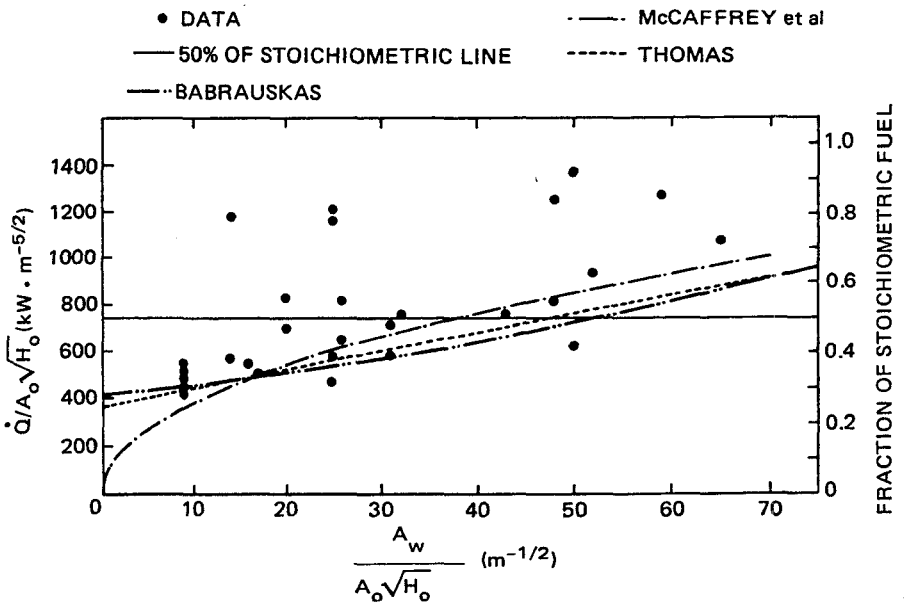

Fig. 3-6.12. The effect of room wall area (gypsum walls) on the heat required for flashover.

$A_0 = 1.08\ m^2$
$H_0 = 1.8\ m$

Therefore

$$\dot{Q} = 610[(0.03)(45.72)(1.08)(\sqrt{1.8})]^{1/2}$$
$$= 860\ kW$$

Method of Thomas

Thomas uses the energy balance for the upper layer shown in Equation 7, where the gas flow rate out of the opening is approximated by[2]

$$\dot{m}_g \approx 0.5A_0\sqrt{H_0} \qquad (47)$$

Thomas develops an expression for $\dot{q}_{loss}$ which assumes the area for the source of radiation for roughly cubical compartments is $A_T/6$

$$\dot{q}_{loss} \approx h_c(T_g - T_w)\frac{A_T}{2} + \varepsilon\sigma(2T_g^4 - T_w^4 - T_{flr}^4)\frac{A_T}{6} \qquad (48)$$

where

A_T = total area of the compartment-enclosing surfaces (m^2)
h_c = convective heat transfer coefficient ($kW/m^2 \cdot K$)
T_w = temperature of the upper walls (K)
T_{flr} = temperature of the floor (K)

From experimental data, Thomas developed an average for $\dot{q}_{loss}$ of $7.8A_T$. Using an upper layer temperature of 577°C or a ΔT_g of 600°C for flashover criterion and $c_p = 1.26\ kJ/kg \cdot K$, yields an expression for the minimum rate of energy release for flashover

$$\dot{Q} = 7.8A_T + 378A_0\sqrt{H_0} \qquad (49)$$

Comparison of Methods for Predicting Flashover

Babrauskas has compared the effect of room wall area on the energy release required for flashover, using the above methods.[19] The results of his comparisons, along with some experimental data for rooms with gypsum board walls, are shown in Figure 3-6.12. The graph shows the energy required for flashover as a function of compartment wall area, both normalized by the ventilation factor

$$A_0\sqrt{H_0}$$

Babrauskas observes that over the range of compartment sizes of most interest, all of the methods produce similar results. The method of McCaffrey *et al* diverts from the others for small room sizes. Babrauskas notes that all of the methods are a conservative representation of the data.

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